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A. C. TRUE, Director.

THE SELECTION AND INSTALLATION OF MACHINERY FOR SMALL PUMPING PLANTS.

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LETTER OF TRANSMITTAL.

U. S. Department of Agriculture,
Office of Experiment Stations,
Washington, D. C., August 1, 1910.

Sir: I have the honor to transmit herewith a report on the selection and installation of machinery for small pumping plants, prepared by Prof. W. B. Gregory, of Tulane University, under the direction of Samuel Fortier, chief of irrigation investigations. In many sections of the United States, and especially in the regions where rice is grown in Louisiana, Texas, and Arkansas, many pumping plants are being installed which are too small to warrant the employment of a consulting engineer. This report is intended to aid in the installation of such plants.

It is recommended that this report be published as a circular of this Office.

Respectfully,

A. C. True,
Director.

Hon. James Wilson,
Secretary of Agriculture.

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THE SELECTION AND INSTALLATION OF MACHINERY FOR SMALL PUMPING PLANTS.

INTRODUCTION.

A considerable area of land in the rice country of Louisiana, Arkansas, and Texas is irrigated by means of water pumped from wells, bayous, or rivers with small steam or gasoline pumping plants capable of supplying the needs of single farms. The capacities of the engines range from less than 10 up to 70 horsepower or more, but in a majority of cases they are between 20 and 50 horsepower. The pumps used in these small plants include air lifts, deep-well, centrifugal, chamber-wheel, and many special forms of pumps. The class of machinery usually bought for these pumping plants is kept in stock by the local dealers. Many of these dealers, however, do not have engineering departments and are able to give only the most general directions regarding the installation of their machinery. The buyers of small plants are therefore largely at the mercy of salesmen, each of whom is ready to declare his particular outfit superior in material, workmanship, and cost of operating to those offered by competitors.

This circular is intended primarily for persons who intend to install plants too small to warrant the employment of a consulting engineer. Some of the principles laid down, however, especially those regarding pumps, are of general application.

The first question confronting the irrigator who must pump his water is, How much water is needed? Knowledge on the following points is necessary to answer this question—acreage to be watered, crop or crops to be raised, nature of the soil, and peculiarities of the climate. For example, if the crop is to be rice, it has been shown that 7.5 gallons of water per acre per minute is required for the heavy clay soils of Louisiana, Texas, and Arkansas, while for the lighter alluvial soils the amount should be increased to 10 or possibly to 12 gallons per acre per minute.

The next fact to be determined is the height through which the water is to be elevated. If a well is to furnish the water, information should be obtained, when possible, from persons having wells in the

same locality, but if it is in new territory pioneer work must be done, and the amount of water to be had and the depth of the water level below the surface of the ground can be known only after the well has been completed and water pumped for a short time. The preliminary pumping is usually done with a temporary pump driven by a traction engine. If the water is to be pumped from a bayou or river, the height through which it is to be elevated can be determined by leveling from the surface of the water in the river or bayou to the height it will stand in the discharge flume or pipe. In all cases the possible lowering of the supply level must be considered and the probable maximum lift determined as accurately as possible. This lift is designated by h in the following formulas.

In the discussion of pumping plants three kinds of horsepower are mentioned—useful water horsepower, brake horsepower, and indicated or rated horsepower. The first named is the amount of power required to raise a given amount of water a given distance in a given time. It is found by one of the following formulas:

$$\text{Useful water horsepower} \begin{cases} = \dfrac{\text{cubic feet of water per second} \times 62.3 \times h}{550} \\ = .113 \times \text{cubic feet of water per second} \times h \\ \text{or} \\ = \dfrac{\text{Gallons of water per minute} \times 8\frac{1}{3} \times h}{33{,}000} \\ = .000252 \times \text{gallons per minute} \times h \end{cases}$$

The brake horsepower (B. H. P.) is the power delivered by an engine to the belt that drives a pump, or to the pump itself in case the unit is direct connected. For steam engines it is usually about 90 per cent of the indicated horsepower and for gasoline engines about 80 per cent. The same idea is expressed also by saying the mechanical efficiency of the engine is 90 or 80 per cent.

The indicated or rated horsepower (I. H. P.) of steam engines is based on the power developed in the cylinder.

This can be determined in an actual case only by means of a steam-engine indicator. However, experiments made by engine builders enable them to tabulate with sufficient accuracy the rated horsepower for engines of given size of cylinder, with a given steam pressure and a given number of revolutions. The practice of all builders is not the same in this particular. Some state that an overload of 25 per cent is allowable, while others give the maximum power the engine is capable of indicating.

Gasoline engines are rated on brake horsepower.

In any case, the basis of rated horsepower ought to be definitely known in order that a fair comparison may be made of different

engines. If an engine to develop a certain horsepower is to be selected on cost consideration only, competition will force dealers to give the smallest machine that will do the work. If, on the other hand, the selection is left entirely to the judgment of the dealer, he may choose a larger engine than is necessary. In this way he will keep on the safe side and also increase his profit. The proper size of engine to select will be somewhere between these two extremes.

The useful water horsepower is usually only a small fraction of the power required at the engine because the efficiency of the pump and of transmission, in case the pump is not direct connected, must be considered, and the lower these efficiencies the greater will be the power that is demanded of the engine. If the water is pumped from a well, the efficiency of the well and strainer must also be included.

It must be remembered also that as soon as a well is pumped the level of the water falls to a point that will depend on the nature of the water-bearing material, the well screen, and similar factors, and that the lift will be considerably greater than the apparent lift when the pump is not running. Many tests of the deep wells of Louisiana and Arkansas show that, if the distance from the height of the discharge to the level of the water in a well when the pump is not running be taken as the lift h, the efficiency of the engine, transmission, and pump is only about 20 per cent. This means that having computed the useful water horsepower by the above formula the indicated horsepower of the engine must be five times as great. If the source of water supply is a bayou or river, the useful water horsepower can be computed as above and the brake horsepower obtained by the following formula:

$$\text{B. H. P.} = \frac{\text{Useful water horsepower}}{\text{Efficiency of transmission} \times \text{efficiency of pump.}}$$

The efficiency of transmission ranges ordinarily from 90 to 95 per cent with its probable value nearer the latter figure for usual practice. The efficiency of small pumps will vary from 30 to 70 per cent, but unless a guaranty of efficiency can be secured it will be safer to use 50 per cent rather than a higher figure as a probable value of pump efficiency. The above formula becomes then

$$\text{B. H. P.} = \frac{\text{Useful water horsepower}}{.93 \times .50} = \text{Useful W. H. P.} \times 2.15$$

In a given case, with the type of pump and its efficiency known, the formula may be corrected for the more definite value of pump efficiency.

PUMPS.

TYPES OF PUMPS.

Seven types of pumps have been used in small plants for irrigation purposes, namely:

PUMPS OPERATED BY THE DIRECT ACTION OF STEAM PRESSURE.

Pumps of this class have two cast-iron vessels into which water is drawn alternately by the condensation of steam. When one vessel is nearly filled a valve is opened and steam pressure is brought on top of the water, driving it out. The steam remaining in the vessel is condensed by a spray of water and a partial vacuum is formed which draws in a fresh charge of water; meanwhile the other vessel is being filled with steam. The opening of the valves is accomplished by means of a small auxiliary steam engine, which forms a part of the pump. This type of pump is excellent for the use of a contractor who desires to drain a pit in a short time, but for pumping for irrigation purposes it is entirely too wasteful of steam. It has been tried out thoroughly in the rice country of Louisiana and Arkansas and has disappeared entirely in the former State and is fast disappearing in the latter.

STEAM-DRIVEN PISTON PUMPS.

Pumps with steam and water pistons attached to opposite ends of a piston rod are familiar as boiler-feed pumps. They are often of the duplex type, so that the movement of each piston rod controls the steam valve of the other. They have free pistons and carry full boiler pressure to the end of the stroke and consequently are wasteful of steam. The economy of operation may be increased by compounding, but this increases the cost of the pump. These pumps are seldom used for irrigation except on a small scale, owing to the small capacity as compared to the centrifugal pump and engine of the same cost and to the excessive cost of operating as compared with better pumps. On the other hand, they are reliable and require little attention.

POWER-DRIVEN PISTON PUMPS.

A belt-driven or direct-connected power pump may be used for irrigation. Power is transmitted from a steam or gasoline engine by a belt or through a train of gears to a piston or pistons. This pump is efficient when properly adjusted, and when three plungers are used the discharge is nearly uniform. It is capable of lifting water against high heads at a reasonable cost for fuel. The objection to its use is the excessive first cost; or, to state it another way, the capacity, when compared with the same outlay for a centrifugal pumping outfit, is so small as to exclude it from irrigation.

DEEP-WELL PUMPS.

These pumps are made by several reliable firms and are used where water must be raised from great depths. They are used where a small quantity of water is required and where this type of pump is the only one that could be employed, with the possible exception of the air lift. The cylinder and piston of the pump are placed deep in the well so that the pump is always primed. The distance below the surface of the ground often exceeds the possible limits of suction lift. Gearing in the "head" of the pump transmits a vertical reciprocating motion to the piston through a long rod. The capacity of the deep-well pump is so limited that a single pump seldom furnishes water to irrigate more than 5 or 10 acres of vegetables, and so far as is known this pump has never been used for rice irrigation.

AIR LIFTS.

These have been used in Arkansas for irrigating rice where the water stands 50 feet or more below the surface of the ground. A fly-wheel compressor is used having steam and air cylinders attached to the same crosshead by means of the piston rods. The compressed air is conducted first to a tank and then piped to the wells to be pumped. The results obtained depend somewhat on the submergence of the air pipe. Although the machine uses the steam expansively, the efficiency of an air lift is inherently so low that the cost of pumping is excessive and it is to be used only when no other means is available.

CHAMBER-WHEELS, OR ROTARY, PUMPS.

Chamber-wheel, or rotary, pumps are usually made with two moving parts that keep in contact with the shell of the pump and with each other as they are revolved. Water is carried in the recesses of the moving parts and is forced out into the discharge pipe of the pump. The workmanship on these pumps is good and a high mechanical efficiency is obtained. One objection to this type of pump is that the flow of water is not uniform, but is alternately accelerated and retarded. Liberal air chambers must be provided on both suction and discharge sides near the pump to take care of the pulsations of the water. These pulsations cause stresses on the pump which tend to spring the shafts and to loosen the impellers and gears on the shafts. All of these difficulties may be avoided by careful design and first-class workmanship and by keeping the speed of operating below the rate at which the pulsations of the water are dangerous. It is practically impossible to increase the capacity of this type of pump beyond its normal rating, because the capacity is directly proportional to the speed of rotation, and to exceed a safe

speed means that the probability of trouble and breakdown is greatly increased. The pump must therefore be of ample capacity for the volume of water required. Another objection to these pumps is that they can not be used where there is a great deal of sand discharged with the water, as the clearances are small, and broken gears, bent shafts, and other troubles will soon develop. The high efficiency of this class of pumps, both large and small, is established beyond question. The cost is high because of the amount of metal and the great amount of machine work required. These pumps are not kept in stock, but must be ordered.

CENTRIFUGAL PUMPS.

This type of pump is used more for irrigating than all the other forms combined. These pumps are made in various forms with horizontal or vertical shafts, depending somewhat upon the nature of the work to be done. They are made with discharge pipes varying in size from 1 inch to 6 feet in diameter.

They are used to elevate water through heights varying from a few feet to several hundred feet. Where the lift is low, slow speed of rotation is used. Single centrifugal pumps driven at high speed by steam turbines may discharge against a head of 180 feet or more, while belted or electrically driven pumps are used often to discharge water against a head of 100 feet. Centrifugal pumps are sometimes compounded, the discharge of one pump leading to the suction of another, arranged usually on the same shaft.

Centrifugal pumps having guide vanes in the passages, after the water leaves the impeller are known as turbine pumps. There is no clear-cut line of difference in conditions under which the turbine pump is used in preference to the ordinary centrifugal, as the reasons for using guide vanes are often mechanical rather than hydraulic. Turbine pumps are seldom used for low lifts, although the multistage turbine pump has been used in special cases for as low lifts as 50 feet, because of unusual conditions. In general the turbine pump is not used for a head less than 75 feet.

The designer of these pumps has three factors to consider—capacity, head pumped against, and speed of rotation. A change in one necessitates changes in the others, and the successful designer is one capable of adjusting these three variables to fit the requirements of a given case and give a reasonably high efficiency. The efficiency of carefully designed turbine pumps does not differ materially from that of ordinary centrifugal pumps.

In general, where either type can be used the horizontal-shaft centrifugal pump is to be preferred to the vertical-shaft pump, as it allows the impeller to be balanced hydraulically, so that the end

thrust of the shaft is eliminated. This is done either by means of double suction pipes, by having a single suction pipe joined to a Y-shaped casting that conducts the water to both sides of the impeller, or by some special balancing device.

Vertical-shaft pumps are used almost exclusively in wells, as there is often a considerable rise and fall in the water level, and a horizontal-shaft pump would have the pulley and belt or rope drive submerged at high water. They are used also in special forms of centrifugal pumps where the lift is not great. Hydraulic balance is desirable, but usually the bearing which supports the moving parts of the pump must carry the weight of shaft, pulley, and impeller, and usually some unbalanced hydraulic pressure. This bearing should be well designed and must be adjusted from time to time to prevent the impeller dragging on the casing.

Vertical-shaft, self-contained pumps, with steel pits, are among the most satisfactory pumps used to elevate water from wells (fig. 1). They are made two-stage for heads up to 100 feet. Vertical-shaft pumps of the old type are liable to settle with the well casing, while the bearing that carries the moving parts remains stationary, or the opposite state of affairs may happen. In either case the impeller rubs against the casing, using much power to overcome the friction and quickly destroying the pump. Wooden pits of the old type are expensive, troublesome to maintain, and if they settle they may throw the shaft out of line and cause endless trouble.

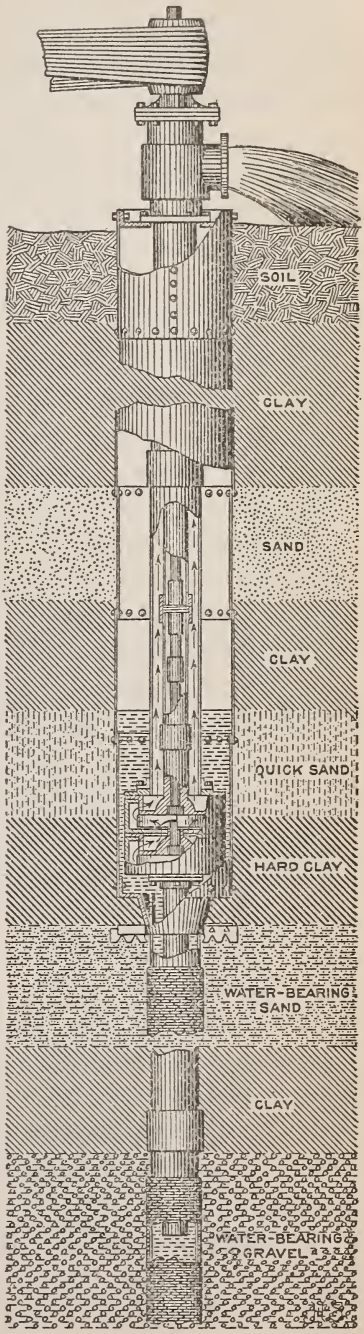

FIG. 1.—Vertical-shaft centrifugal pump with steel pit.

Centrifugal pumps having discharge pipes 12 inches or more in diameter should operate under favorable circumstances with a mechanical efficiency of 60 to 75 per cent when the lift equals or exceeds 10 feet. Many manufacturers are willing to guarantee such efficiencies. Recent tests of several large centrifugal pumps

have shown efficiencies of approximately 80 per cent. Inasmuch as a high efficiency means a smaller engine horsepower than for a low efficiency, and this in turn means less fuel consumed to perform a given amount of work, it is desirable to have an efficiency guaranty, insuring a high-grade pump. The fulfillment of a guaranty can only be shown by a careful test. The quantity of water pumped must be measured either by means of a weir, used under conditions where its accuracy is known, or by some other standard instrument. The head pumped against and the power delivered to the pump must be measured. Such a test can only be made by an expert who has the necessary instruments and who is experienced in using them.

DYNAMIC HEAD, OR "HEAD ON PUMP."

Specifications for pumps and guaranties of pump efficiencies should state very clearly what is meant by "head." Failure to define this term accurately has led to many disappointments. A pump may be required to raise a certain quantity of water through a known difference of level, and in this case the designer must know the length and arrangement of the suction and discharge pipes needed, so that he may figure the "head on pump," which includes all necessary losses in the piping system in addition to the actual difference of level through which the water is to be elevated. Sometimes the head on pump, or "dynamic head," is specified, and this includes all losses due to friction in the suction and discharge pipes, the loss due to the water entering the suction pipe, and the energy of motion in the water as it is discharged, as well as the actual lift.

The head due to actual lift is determined by measuring the vertical distance from the suction to the discharge level. The head on pump, or dynamic head, must be obtained by computation in designing a pumping plant or obtained experimentally in a plant already in operation.

To determine head on pump experimentally it is necessary to have openings in the suction and discharge pipes, near the pump, so that the pressures may be read at those points (figs. 2 and 3). The quantity of water discharged must also be known at the time the pressure observations are taken, so that the mean velocity of water in the suction and discharge pipes may be known. The total head on pump, or dynamic head, H, is obtained by the following formula:

$$H = h_d - h_s + h^1 + \frac{V_d^2 - V_s^2}{2g}$$

in which

h_d = pressure on discharge pipe measured in feet of water.

h_s = pressure shown on suction pipe measured in feet of water.

h^1 = vertical height in feet between points where h_s and h_d are measured.

V_s=velocity in suction pipe in feet per second.

V_d=velocity in discharge pipe in feet per second.

g=acceleration of gravity=32.2 feet per second, per second.

If the pump is above the water, which is usually the case, h_s is negative.[a] If it is submerged, as the vertical-shaft pump shown in figure 3, page 16, it is positive. If the suction and discharge pipes are of the same diameter at the sections where h_s and h_d are measured, $V_d=V_s$ and the $\frac{V_d^2-V_s^2}{2g}$ becomes zero. Again, if the two points at which pressures are obtained are at the same level, h^1 equals zero.

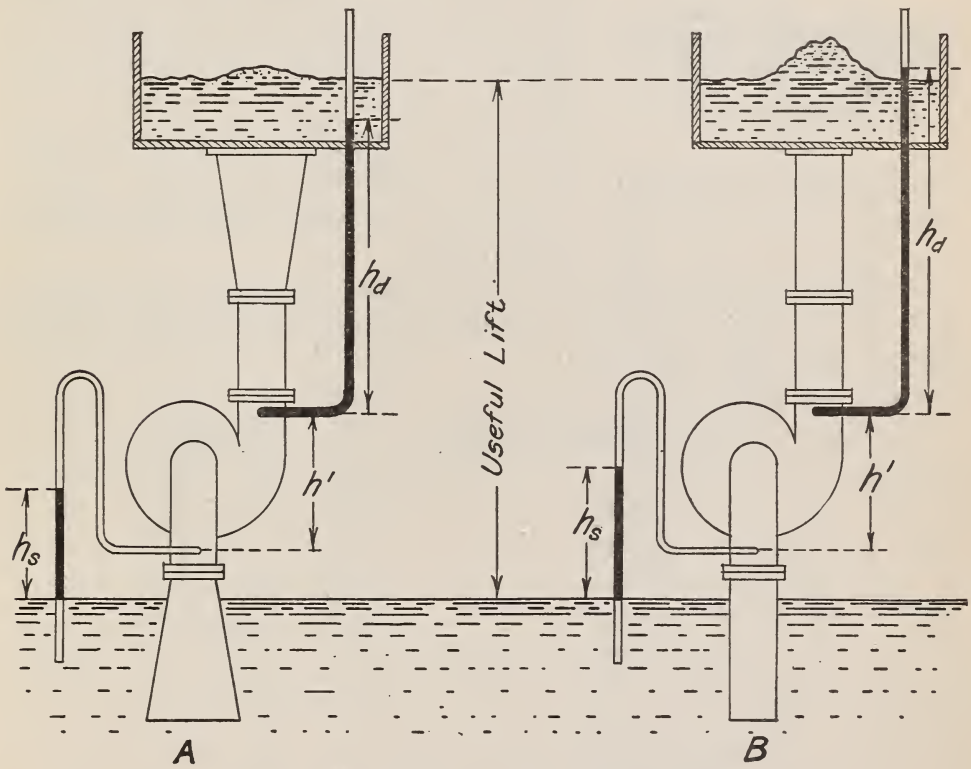

FIG. 2.—Methods of measuring head on pump.

The above statement for the value of H includes all losses in the piping system and gives a method of comparing pumps on their merits, regardless of peculiarities of piping and local conditions.

To obtain h_s in well pumps the suction pipe should be tapped, as shown in figure 4, and the gage pipe carried to the top of the well, where a vacuum gage may be attached to measure the suction head [b]

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[a] If h_s is negative, $H=h_d-(-h_s)+h^1+\frac{V_d^2-V_s^2}{2g}$

or $H=h_d+h_s+h^1+\frac{V_d^2-V_s^2}{2g}$

[b] Vacuum gages usually read in inches of mercury. To reduce inches of mercury to feet of water, multiply by 1.13.

and to give an indication from which the proper performance of the pump, strainer, and well may be judged.

With a long suction pipe extending into a well, as shown in figure 4, and with the water level below the pump, the conditions are somewhat similar to those shown in figures 2 and 3, pages 15, 16.

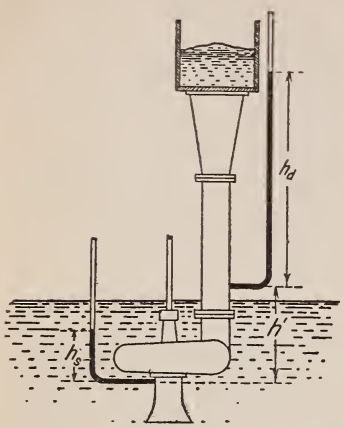

FIG. 3.—Method of measuring head on pump for submerged pump.

The dotted horizontal line in figure 4 shows normal height to which the water would rise when not pumping from the well. The two dotted lines curving downward show the way in which the water level falls near the casing when the pump is in operation. The suction head will be a negative head or vacuum, and if there is air and not water in the small pipe the indication will be the vacuum at the level where the small pipe taps the suction pipe.

It will be seen from the two arrangements of piping (fig. 2, p. 15) that the total head for A is considerably less than that for B. The reason for this will be brought out later.

Sometimes useful water horsepower is referred to in the test of a pump. This is computed by one of the formulas given on page 8, using for the head or lift h, the actual height through which the water is elevated. When the water horsepower is computed on the work done by the pump, the head, h, in the formula is changed to H, as defined above.

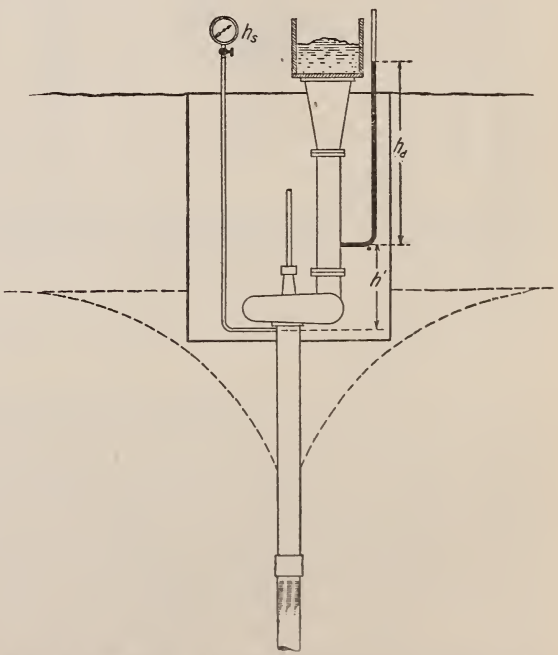

FIG. 4.—Method of measuring head on pump for pump in well.

LOSSES IN PUMP INSTALLATIONS.

Long suction and discharge pipes and sharp bends are to be avoided when possible, to lessen the cost of installation and to prevent the friction losses in pumping. The latter are

fairly well known as the result of experiments. They are expressed by the formula $h_f = f \frac{L}{D} \frac{V^2}{2g}$ in which

f is a coefficient, usually equal to .02.

L=length of pipe in feet.

D=diameter of pipe in feet.

V=mean velocity of water in pipe in feet per second.

g=the acceleration of gravity=32.2 feet per second, per second.

The friction losses in feet of water for various lengths of pipe, 12 inches in diameter, and for three different velocities are given in the following table:

Loss of head due to friction of water flowing at different mean velocities through 12-inch pipes of different lengths, mean velocity per second.

Length of pipe.	Loss of head.		
	8 feet velocity.	10 feet velocity.	12 feet velocity.
Feet.	*Foot.*	*Feet.*	*Feet.*
5	0.10	0.15	0.22
10	.20	.31	.45
15	.30	.46	.67
20	.40	.62	.90
30	.60	.93	1.34
40	.80	1.24	1.79

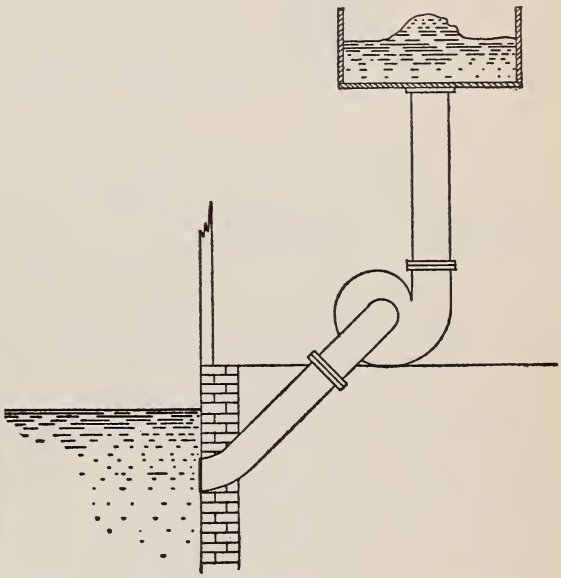

FIG. 5.—A wrong method of installing a pump.

The intake ends of suction pipes should be enlarged, because the loss at entrance to the suction pipe is $0.5\frac{V_s^2}{2g}$ for conditions shown in figure 5 and may be as much as $0.93\frac{V_s^2}{2g}$ for conditions illustrated in figure 2, B, or figure 6, pages 15, 18. By increasing the area of the pipe at intake the velocity is reduced and the loss, which varies as the square of the velocity, is greatly reduced. In the examples that follow the smaller value is used, and although it can not be claimed that the conditions of figure 5 are prevalent, the results are conservative, because they represent the minimum saving that may be effected.

Supposing the velocity in the 12-inch pipe to be 10 feet per second, so that $0.785 \times 10 \times 7.5 \times 60 = 3{,}530$ gallons per minute are pumped, the loss of head at entrance will be $0.5 \times \frac{10^2}{64.4} = 0.78$ foot. If a conical pipe, like that shown in figure 2, A, is substituted for the 12-inch pipe,

the opening being 15 inches, the loss is reduced to 0.32 foot of water. If enlarged to 18 inches the loss is only 0.15 foot.

It is important also to enlarge the discharge pipe, because the energy of motion that remains in the water is lost, and a large cross section means a low velocity. It is not unusual to see water leaping into the air from the discharge pipe of a pump giving a fountain effect on a large scale. This ought not to be, as it is a waste of fuel and money. The energy thrown away is proportioned to the square of the final velocity of the water discharged, and therefore the velocity ought to be kept low. The loss of head for a 12-inch pipe, with no enlargement at the end for a velocity of 10 feet per second, would be $\frac{10^2}{64.4}=1.55$ feet of water. If an enlarged pipe is used, expanding from 12 to 15 inches, the loss would be 0.64 foot of water, and if enlarged to 18 inches the loss would be only 0.31 foot of water.

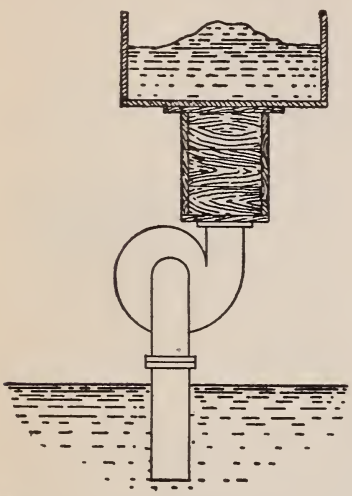

FIG. 6.—Another wrong method of installing a pump.

The following table has been computed to show the gains, both in feet and per cent, resulting from enlarging the ends of the suction and discharge pipes of a 12-inch centrifugal pump to 15, 18, and 24 inches. The actual lift is 15 feet and the friction loss 0.46 foot in each case.

Decrease of loss of head at suction intake, loss due to discharge velocity, friction head, and total head on the pump, due to enlarging the ends of the suction and discharge pipes.

Kind of pipe.	Loss of head at intake $\frac{0.5V_s^2}{2g.}$	Loss of head at discharge $\frac{V_d^2}{2g.}$	Friction loss in 15 feet of pipe.	Total head to be produced by pump.	Gain.	
	Foot.	*Feet.*	*Foot.*	*Feet.*	*Feet.*	*Per cent.*
12-inch straight pipe.....	0.78	1.55	0.46	17.79	0	0
12-inch enlarged to 15-inch	.32	.64	.46	16.42	1.37	7.7
12-inch enlarged to 18-inch	.15	.31	.46	15.92	1.87	10.5
12-inch enlarged to 24-inch	.05	.10	.46	15.61	2.18	12.2

If the same velocity, 10 feet per second, is maintained in the discharge nozzle of the pump, the actual lift may be changed, but losses at the intake and discharge will not change. The loss of head due to friction in pipe is arbitrarily assumed to be the loss that would result

in a length of pipe equal in length to the actual static lift or difference of suction and discharge levels. This will, in general, be practically correct. The total head on the pump is the head that must be produced by the pump in order to maintain a velocity of 10 feet per second in the discharge nozzle, for the various conditions stated. With a larger actual lift than that used in the above example, the losses enumerated will be relatively less, while for an actual lift less than 15 feet, these losses will be relatively greater. The tabulated results obtained by the above method for velocities in pipes of 8, 10, and 12 feet per second, have been platted in figures 7, 8, and 9. A glance at these curves will show the importance of properly designed pipes to conduct water to and from a pump. The enlargement of the pipes should not be abrupt, as in figure 6, page 18, but gradual, as in figure 2 A, page 15. A rule for enlarging the ends of suction and discharge pipes that has worked satisfactorily is to have the diameter of the end $1\frac{3}{4}$ that of the pipe and to begin the enlargement a distance from the end equal to $2\frac{1}{2}$ times the diameter (fig. 10). In some cases the discharge pipe will need to be flanged on both ends. It is convenient also, in case a flap valve is used to close the end of the discharge pipe while the pump is being primed, to change the section of the pipe from circular to square or rectangular. The change does not affect the results, provided the ratio of areas at the two ends of the pipe is maintained.

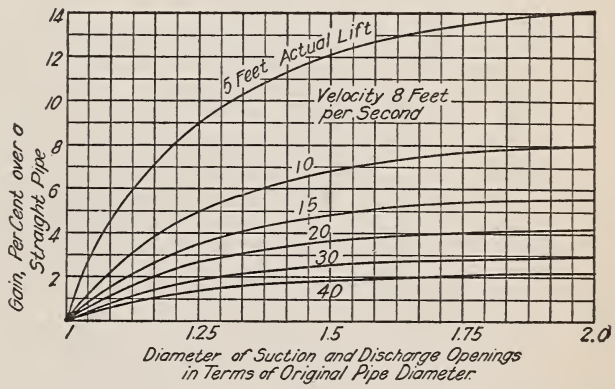

FIG. 7.—Decrease of losses due to enlarging ends of pipe. Velocity 8 feet per second.

In case a well-designed pumping plant is to be erected and the velocity of the water in the suction and discharge pipe is to be as low as 8 feet per second if a 12-inch pipe is used the question arises whether it will pay to use a larger pipe, say 18 inches in diameter, at an increased cost, thereby reducing the fuel cost of operating each season. An analysis of the problem, using average conditions, shows that the 12-inch pipe is the better proposition so long as the velocity in this pipe is not more than 8 feet per second and the cost of fuel to irrigate an acre not more than $1.50 per season. This necessitates an economical plant, much better than many of the small ones now in use, and calls for a carefully designed plant.

If the velocity is increased to 10 feet per second, the 18-inch pipe is the better proposition for general conditions and will save a neat sum in wasteful plants. The larger pipe should be used when conditions make an unusual length of pipe necessary. The greater the length of pipe in proportion to the actual lift the greater the gain by keeping the velocity low and so reducing friction. Whether or not enlarged suction and discharge pipes are used, the expanded ends should be employed to avoid the losses at entrance and exit. The cost per linear foot of spiral riveted pipe of varying diameters is shown in figure 11.

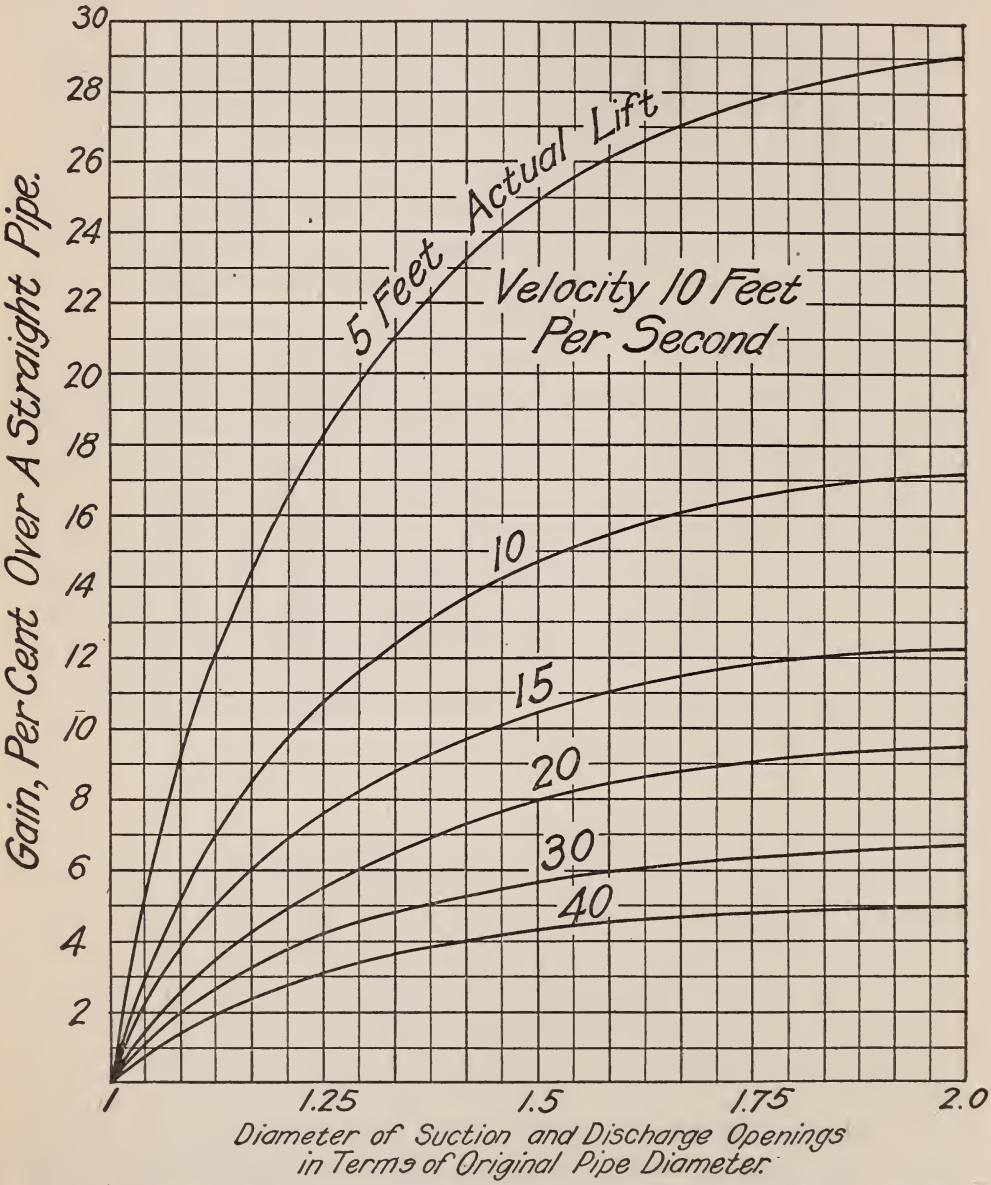

FIG. 8.—Decrease of losses due to enlarging ends of pipe. Velocity 10 feet per second.

A foot valve should never be used on a pump supplying irrigation water on account of the excessive loss of head which it causes. A

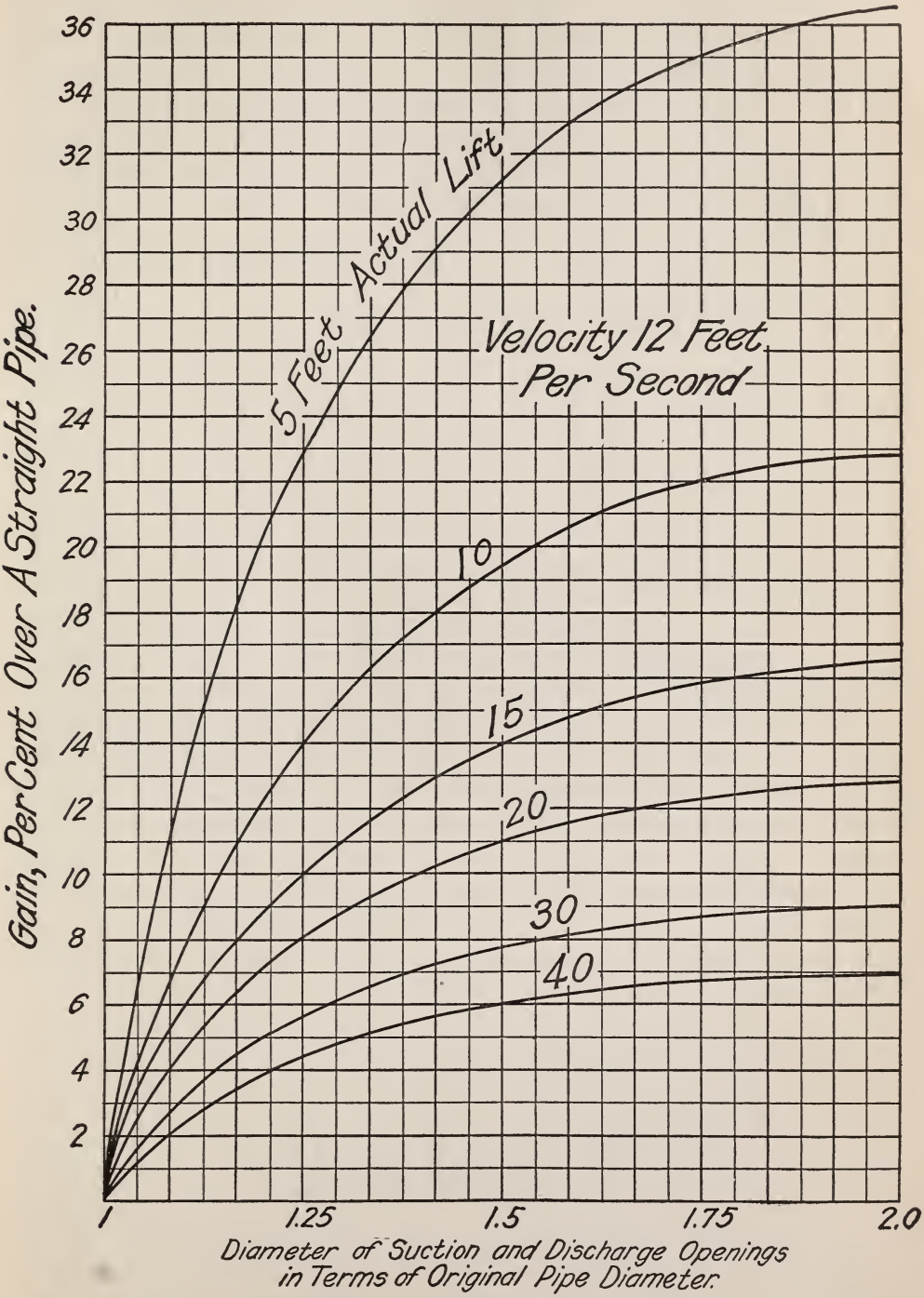

FIG. 9.—Decrease of losses due to enlarging ends of pipe. Velocity 12 feet per second.

screen should be provided in the suction canal to prevent trash from reaching the pump. The area of such a screen must be large so that

the velocity of the water will be small in order to prevent loss of head.

The importance of avoiding unnecessary losses is realized when it is remembered that any decrease in total head required of the pump will decrease the demand for power from the engine. In any case, this will mean easier running for the engine and less fuel demanded. In a steam plant it will often mean a more economical load for the boiler as well as less fuel to purchase and to handle.

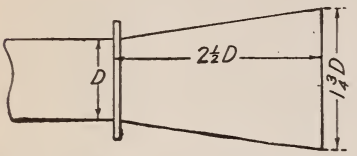

FIG. 10.—Method of enlarging ends of discharge and suction pipes.

FOUNDATIONS FOR PUMPS.

When a pump is direct-connected to the engine driving it, care must be exercised in designing and building the foundations. It is best to use a flexible coupling in such a case so that a slight difference in alignment will not cause undue wear or lead to the destruction of the machinery. Where a belt or rope drive is used the design of foundations will depend largely upon the type of pump and on local conditions.

COST OF CENTRIFUGAL PUMPS.

The prices given below are for a well-known make of centrifugal pump. A steam injector for priming and a flap valve for the end of the discharge pipe, as well as suction and discharge pipe and freight should be added to give total cost.

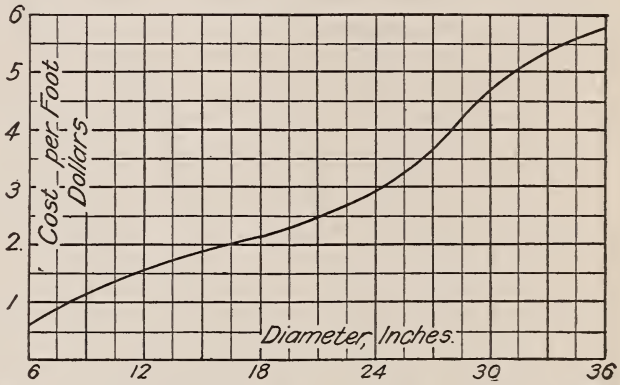

FIG. 11.—Diagram showing cost of spiral rivet galvanized iron pipe of different diameters.

Cost and size of centrifugal pumps used under varying conditions.

Size.	Cost.	
	Low lifts, high speed.	Medium lifts, medium speed.
Inch.		
6	$119	$150
8	195	263
10	247	345
12	328	586
14	375	654
16	473	825
18	890	962
20	990	1,100
24	1,179	1,314

ENGINES.

TYPES OF ENGINES.

For small irrigating plants steam or gasoline engines are used. In some sections one type is used almost exclusively, while in other sections the other type prevails. Each has its advantages and its disadvantages. The small steam plant requires the constant attention of an operator, which makes necessary the employment of two men if the plant is to be run day and night. On the other hand, small gasoline engines properly adjusted will run for hours with a minimum of attention. If the operator is not skilled in running such an engine, however, it may give a great deal of trouble.

The steam plant is flexible to a certain extent, because the engine will respond to a rise in steam pressure and will carry an overload if carefully handled, while the maximum load of a gasoline engine can not be exceeded. Gasoline engines are designed usually to carry about 15 per cent greater rating than the nominal horsepower. An attempt to run them beyond their true maximum load will result in a slowing down of the engine.

The choice between a steam or gasoline outfit is sometimes settled by personal opinion. If a man has had experience in operating one or the other and understands the difficulties that are liable to arise, while the other type of plant is unknown to him, first cost and operating expenses being fairly well balanced, he will decide upon the plant with which he is familiar.

The design of gasoline engines is fairly well standardized. There are, however, differences in details of design, materials used, and finish. In general the prices of these engines are an index to quality. In the smaller steam plants the cheapest grade of slide-valve, noncondensing engines has been used to the exclusion of all others. Whether this was a wise choice depends upon circumstances, some of which will be discussed briefly. The small noncondensing steam engines employ either a slide or piston valve, the latter being generally found on automatic engines with shaft governors. The cheaper engines have throttling governors and maintain a sufficiently close speed regulation for this class of work. The workmanship of these engines differs greatly. Some are made carelessly with no attempt at proper steam distribution, while others are designed so that they are satisfactory, both in the use of steam and in mechanical operation.

The two methods of governing a simple engine may be illustrated by the indicator cards A and B, figure 12. Card A would be obtained by means of a steam-engine indicator from an automatic engine in which the governor controls the speed by changing the point of cut-off corresponding to the change of load. The initial pressure of the steam remains practically constant. The point of compression is

changed ordinarily but little, and if a separate exhaust valve is used it is fixed. Card B is such as would be obtained from a throttling engine. The point of cut-off is fixed, but the governor reduces the pressure of the whole card by throttling. Both cards show fairly high compression pressure and would be suitable for reasonably high speeds, because the moving parts of the engine would be cushioned by the compressed steam and so brought to rest at the end of the stroke without shock.

Indicator cards like C, figure 12, are to be had only from engines of improper design. The cut-off is so late that the steam is given very little opportunity to expand. As practically no compression takes place, such engines can be operated only at low speeds without excessive pounding, which contributes to rapid wear and besides is disagreeable.

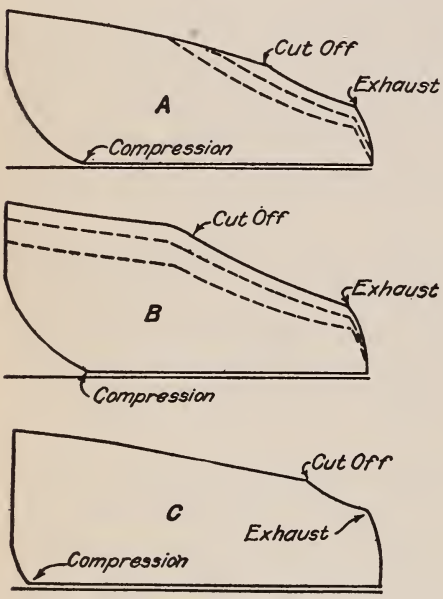

FIG. 12.—Indicator cards of different types of engines.

A few engines are made with adjustable cut-offs so that the speed may be varied by changing the point of cut-off while the engine is running. A throttling governor is provided to prevent the engine from running away in case the pump loses its priming. This device is a valuable addition to an engine where the level of suction water varies considerably during a season, which necessitates that the revolutions of pump and engine be varied accordingly.

Compound engines are more economical of fuel than simple engines, as they use the steam through a greater ratio of expansion. A compound engine may be run condensing or noncondensing. In either case it is more complicated than the simple engine and requires greater skill to operate successfully and keep in adjustment, although a man who can operate a simple engine successfully should have no trouble in operating a compound engine if proper care is exercised. The increased economy of the compound engine over the simple engine must be balanced against the greater cost of the compound. The cost of a condensing engine and accessories will be greater than the cost of the noncondensing outfit.

The desirability of using a condenser will depend on circumstances. There is always plenty of water around a pumping plant, and sometimes a barometric condenser may be installed, so that no extra pumps are required. In such a case the water is started flowing through the

condenser by means of a steam ejector. A jet condenser and vacuum pump may be used to advantage under some conditions.

Compound engines may be had down to 15 horsepower, although only a few builders make the smaller sizes. Several firms furnish them with capacities of 50 horsepower or more. The steam pressure used in the smaller plants rarely exceeds 100 pounds per square inch at the boiler. This is the standard pressure for small boilers of reasonable cost, and the figures given below are based on a boiler pressure of 100 pounds. Steam of higher pressure may be used advantageously in compound engines, although the cost of boilers for high-pressure steam will, of course, be somewhat increased.

THE CHOOSING OF AN ENGINE.

It has been shown how the brake horsepower of the engine may be computed where the quantity of water required and the head through which it must be pumped are known. The next step is to choose an engine. In order to do this wisely the cost of operating the various engines, as well as the first cost of the outfit, must be examined. The depreciation, probable repairs, and labor involved in operating a plant must be considered also. The cost of the boiler and accessories must be considered with the engine, because the simple engine uses more steam than the compound noncondensing, and therefore requires a large boiler, while the latter uses more steam and requires a larger boiler than the compound condensing engine. Any attempt to state the steam consumption of a class of engines, even if it applies to the average case, can be seriously objected to if individual cases are considered. In the table on page 26 the steam consumption given is for engines and necessary pumps. It is the amount of steam to be furnished by the boilers.

Again, a statement of costs for engines and boilers may be too high or too low for individual cases because the various makes of engines vary greatly in workmanship and finish. In the discussion that follows the figures for steam rate are taken largely from tests of engines under conditions paralleling those in the plant to be designed. The prices of engines, boilers, and accessories were obtained from several sources, and represent average conditions as nearly as it was possible to arrive at correct figures from many actual quotations. The cost of condensers is estimated at $4, and that of piping and foundations at $3 per brake horsepower for steam engines and $1.50 for gasoline engines. It must be kept in mind that the following discussion refers to the cost of the source of power in a pumping plant and not to the cost of the entire plant. The gasoline engine, with its foundations, etc., is considered and compared with steam engines of various types, with their boilers, accessory machinery, piping, and foundations.

Steam consumption per brake horsepower hour for different types of engines and their accessories.

	Simple.	Compound noncondensing.	Compound condensing.
Steam rate per brake horsepower per hour.....pounds..	50	30	25
Temperature of feed water (a heater is used).....°F..	190	190	140
Thermal[a] units per hour, steam at 100 pounds gauge.....	51,500	30,900	26,900
Thermal units in fuel per hour, boiler efficiency 65 per cent...	79,200	47,500	41,400
Coal per brake horsepower per hour[b].....pounds..	5.87	3.52	3.07
Cost of fuel per brake horsepower hour, at $4 per ton....cents..	1.17	.704	.614
Same, at $6 per ton .....do....	1.76	1.06	.921

[a] Pounds of steam per brake horsepower per hour × factor of evaporation × latent heat of steam from and at 212° F. A British thermal unit is the amount of heat required to raise the temperature of 1 pound of water 1 degree Fahrenheit.

[b] Heat units assumed, 13,500 per pound of coal.

Corresponding boiler horsepower of engines of different types and sizes.

Size of engine.	Type of engine.		
	Simple.	Compound noncondensing.	Compound condensing.
Brake horsepower.	*Boiler horsepower.*	*Boiler horsepower.*	*Boiler horsepower.*
30	47	28	25
50	78	47	40
70	109	65	57
100	155	93	81

The saving in boiler capacity, and therefore in the cost of boiler, is shown clearly in the above table.

It is assumed that each pumping plant will run eighty days per season, twenty-four hours per day, or one thousand nine hundred and twenty hours. The cost of fuel required in the following tables has been figured on this basis. Depreciation is assumed at 10 per cent and interest at 7 per cent of the first cost. The labor required per season is estimated at three months.

A pumping plant having a simple engine or a gasoline engine will require two men, one for the day and one for the night, the minimum wages of the men being $30 per month. The assumed cost of labor for the compound engines, whether condensing or not, is double that for the gasoline or simple engines, as it is difficult to find men who have had experience with compound engines. The estimate of $60 per month for each man required to run one of these plants is conservative. The cost of labor is taken at a low figure because small pumping plants are often operated by their owners, at least in part, and ordinary farm labor is utilized also for this purpose. In case the labor is employed by the year, the figures assumed above for cost of labor are about right, but the cost would have to be increased if the men were employed only during the pumping season. It is as-

sumed also that the cost of labor will be the same in each plant having the same type of engine, as the pumping plants under consideration are small, requiring 30 to 100 brake horsepower.

Cost of fuel for a season of eighty days for engines of different sizes and types.

[1920 × cost per brake horsepower hour × brake horsepower of engine.]

Brake horsepower of engine.	Type of engine.							
	Simple.		Compound noncondensing.		Compound condensing.		Gasoline.	
	Coal per ton.		Coal per ton.		Coal per ton.		Gasoline, per gallon.[a]	
	$4	$6	$4	$6	$4	$6	10 cents.	15 cents.
30	$674	$1,011	$406	$609	$354	$531	$720	$1,080
50	1,124	1,686	677	1,015	590	885	1,200	1,800
70	1,575	2,362	948	1,422	825	1,237	1,680	2,520
100	2,248	3,372	1,350	2,025	1,180	1,770	.....	.....

[a] The consumption per brake horsepower is assumed to be one-eighth gallon.

If fuel oil is used, $1.15 may be substituted for $4 and $1.72 per barrel for $6 in the above table, assuming 3.5 barrels of oil to be equivalent to 1 ton of coal.

First cost and cost of operating for season of eighty days for pumping outfits of different sizes and types.

30-BRAKE HORESEPOWER OUTFIT.

	Type of engine.			
	Simple.	Compound noncondensing.	Compound condensing.	Gasoline.
Size of boiler.....horsepower..	47	28	25	.....
Cost of boiler	$580	$375	$330	.....
Cost of engine	320	780	780	$1,050
Piping and foundation	90	90	90	45
Condenser	.....	.....	120	.....
Feed pump	25	25	25	.....
Heater	84	84	84	.....
Total first cost	1,099	1,354	1,429	1,095
Interest and depreciation, 17 per cent	187	230	243	186
Labor	180	360	360	180
Fuel at $4 per ton	674	406	354	[a]720
Fuel at $6 per ton	1,011	609	531	[b]1,080
Total cost of operating per season:[c]				
Fuel at $4 per ton	1,041	996	957	[a]1,086
Fuel at $6 per ton	1,378	1,199	1,134	[b]1,446

[a] Gasoline at 10 cents per gallon.
[b] Gasoline at 15 cents per gallon.
[c] Total cost of operation consists of interest and depreciation, labor, and cost of fuel.

First cost and cost of operating for season of eighty days for pumping outfits of different sizes and types—Continued.

50-BRAKE HORSEPOWER OUTFIT.

	Type of engine.			
	Simple.	Compound noncondensing.	Compound condensing.	Gasoline.
Size of boiler.....horsepower..	80	45	40	.....
Cost of boiler	$800	$510	$435	.....
Cost of engine	475	1,150	1,150	$1,625
Piping and foundations	150	150	150	75
Condenser	.....	.....	200	.....
Feed pump	45	25	25	.....
Heater	84	84	84	.....
Total first cost	1,554	1,919	2,044	1,700
Interest and depreciation, 17 per cent	264	326	347	289
Labor	180	360	360	180
Fuel at $4 per ton	1,124	677	590	[a] 1,200
Fuel at $6 per ton	1,686	1,015	885	[b] 1,800
Total cost of operation per season: [c]				
Fuel at $4 per ton	1,568	1,363	1,297	[a] 1,669
Fuel at $6 per ton	2,130	1,701	1,592	[b] 2,269

70-BRAKE HORSEPOWER OUTFIT.

Size of boiler.....horsepower..	110	65	60	.....
Cost of boiler	$1,030	$675	$620	.....
Cost of engine	590	1,400	1,400	$2,400
Piping and foundations	210	210	210	105
Condenser	.....	.....	280	.....
Feed pump	45	25	25	.....
Heater	110	110	110	.....
Total first cost	1,985	2,420	2,645	2,505
Interest and depreciation, 17 per cent	337	411	450	426
Labor	180	360	360	180
Fuel at $4 per ton	1,575	948	825	[a] 1,680
Fuel at $6 per ton	2,362	1,422	1,237	[b] 2,520
Total cost of operation per season: [c]				
Fuel at $4 per ton	2,092	1,719	1,635	[a] 2,286
Fuel at $6 per ton	2,879	2,193	2,047	[b] 3,126

100-BRAKE HORSEPOWER OUTFIT.

Size of boiler.....horsepower..	155	100	80	.....
Cost of boiler	$1,500	$1,000	$800	.....
Cost of engine	740	1,890	1,890	.....
Piping and foundations	300	300	300	.....
Condenser	.....	.....	400	.....
Feed pump	45	45	45	.....
Heater	110	110	110	.....
Total first cost	2,695	3,345	3,545	.....
Interest and depreciation, 17 per cent	458	569	603	.....
Labor	180	360	360	.....
Fuel at $4 per ton	2,248	1,350	1,180	.....
Fuel at $6 per ton	3,372	2,025	1,770	.....
Total cost of operation per season: [c]				
Fuel at $4 per ton	2,886	2,279	2,143	.....
Fuel at $6 per ton	4,010	2,954	2,733	.....

[a] Gasoline at 10 cents per gallon.
[b] Gasoline at 15 cents per gallon.
[c] Total cost of operation consists of interest and depreciation, labor, and cost of fuel.

The relationship between the costs of the various sources of power is shown graphically in figure 13. The relative costs of the different

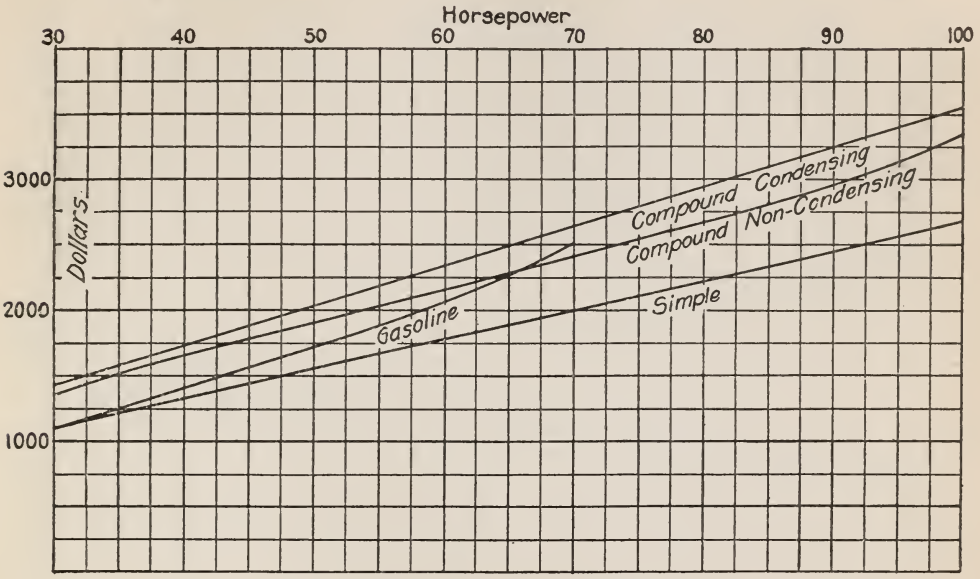

FIG. 13.—Cost of engines, boilers, and accessories of different types and sizes.

types of engines is shown in figure 14 and the relative cost of the boilers and accessories for the different types of engines is shown in figure 15.

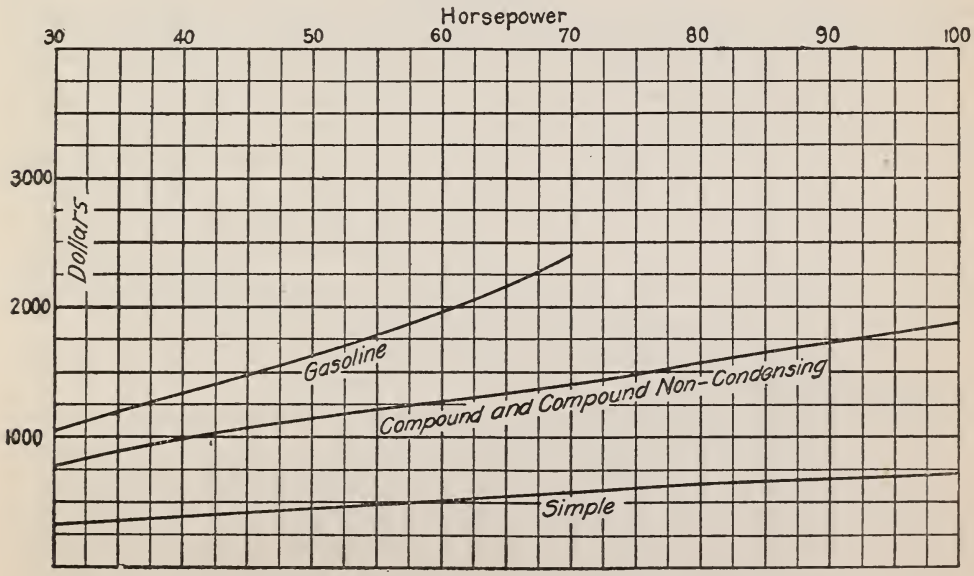

FIG. 14.—Cost of engines of different types and sizes.

With the simple engine the cost of the boiler is considerably greater than for the other two steam plants, but the cost of the engine is so low that the total cost of the outfit is lower for the simple engine.

At 30 brake horsepower the cost of a gasoline engine is practically the same as that of the simple engine. For sizes below 30 horsepower

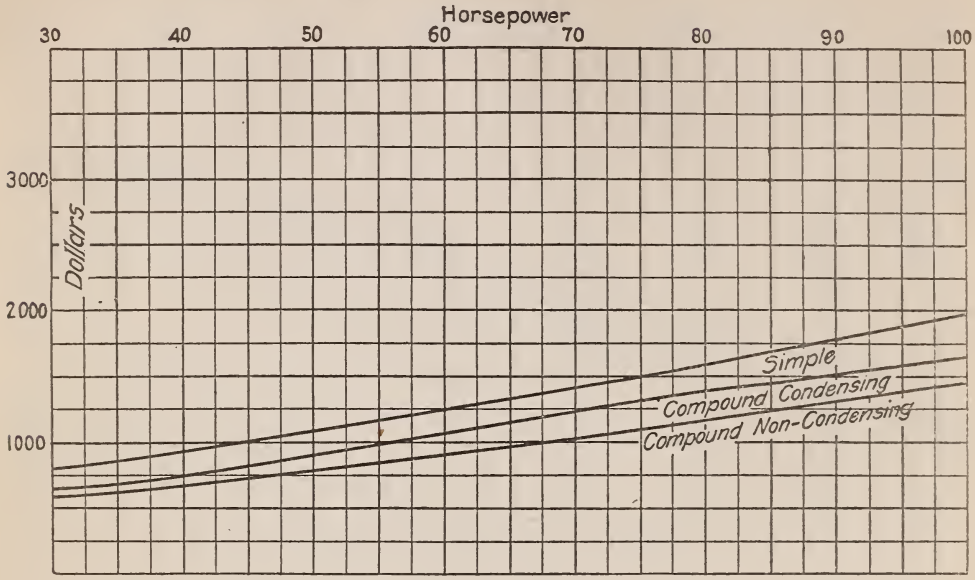

FIG. 15.—Cost of boilers and accessories for engines of different types and sizes.

the price would be as low as the steam plant and probably a little less. As the horsepower increases above 30 the price increases more

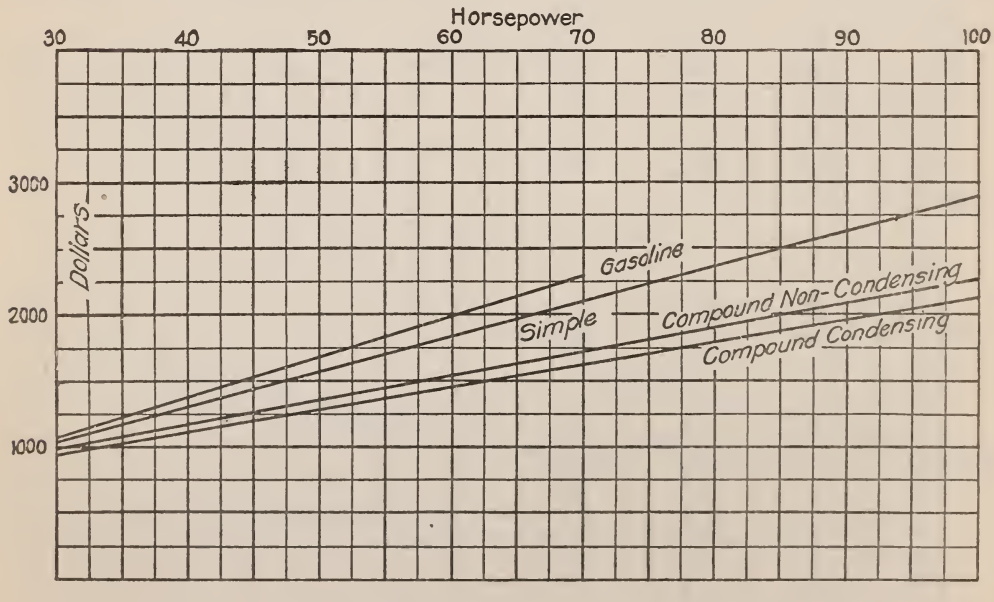

FIG. 16.—Cost of operating plants—coal $4 per ton, or gasoline 10 cents per gallon.

rapidly than that of the simple steam plant and at 70-brake horsepower it is about the same as the compound steam engines. Very few gasoline engines are made larger than 70 horsepower.

The oil engine has been suggested as a prime mover for such pumping plants. These engines are reliable and economical, but the first cost is prohibitive, as it exceeds by more than 50 per cent the cost of

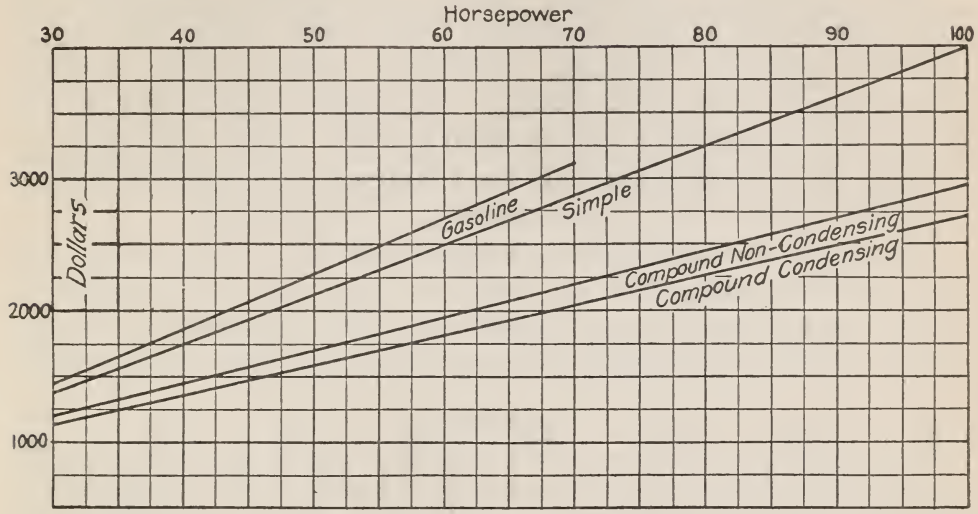

FIG. 17.—Cost of operating plants—coal $6 per ton, or gasoline 15 cents per gallon.

a compound condensing engine, boiler, and accessories, and exceeds by 100 per cent the cost of a simple engine for the smaller sizes, and the increase in cost for larger units is proportionately greater. Fig-

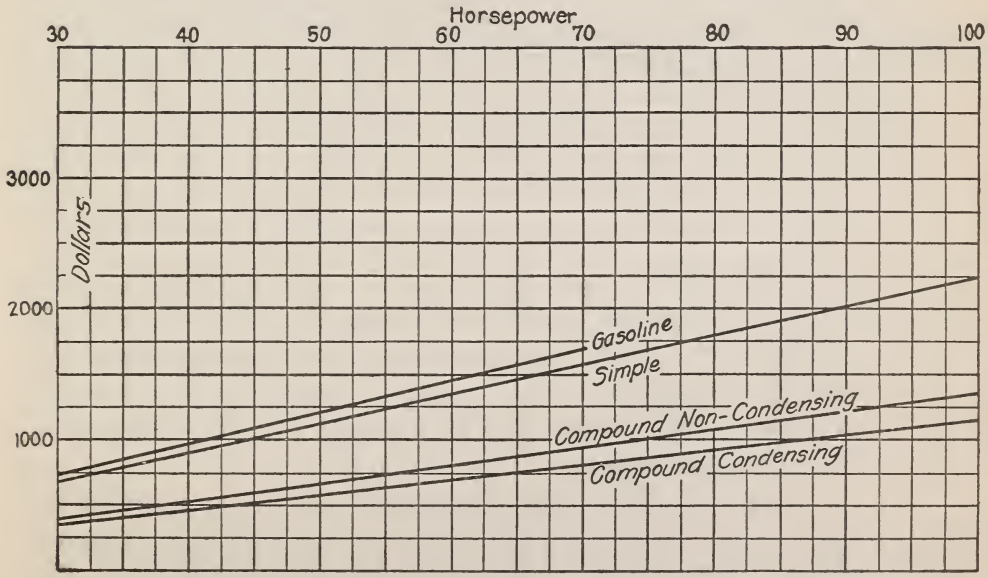

FIG. 18.—Cost of fuel per plant—coal $4 per ton, or gasoline 10 cents per gallon.

ures 16 to 19 show the relative total cost, cost of fuel, including interest on investment at 7 per cent and depreciation at 10 per cent and cost of labor for a season of eighty days for the different types and

sizes of engines. The cost of fuel oil, at $1.15 per barrel, is equivalent to coal at $4 per ton, and the cost of fuel oil, at $1.72 per barrel, is equivalent to coal at $6 per ton.

The method of comparison shown is of general application, but the figures given are not applicable to all cases, as the cost of fuel varies for different localities. Fuel oil is used in small pumping plants in some parts of Louisiana. The cost is as great as $1.40 per barrel, the equivalent of coal at $4.90 per ton. Again, the prices of engines, boilers, and accessories are averages, and cheaper and more expensive engines than those selected may be purchased.

With costs of machinery as assumed and fuel costing $4 per ton for bituminous coal, $1.15 per barrel for fuel oil, and 10 cents per gallon for gasoline, figure 16 shows very little difference in the cost per season of the four plants under consideration when the developed

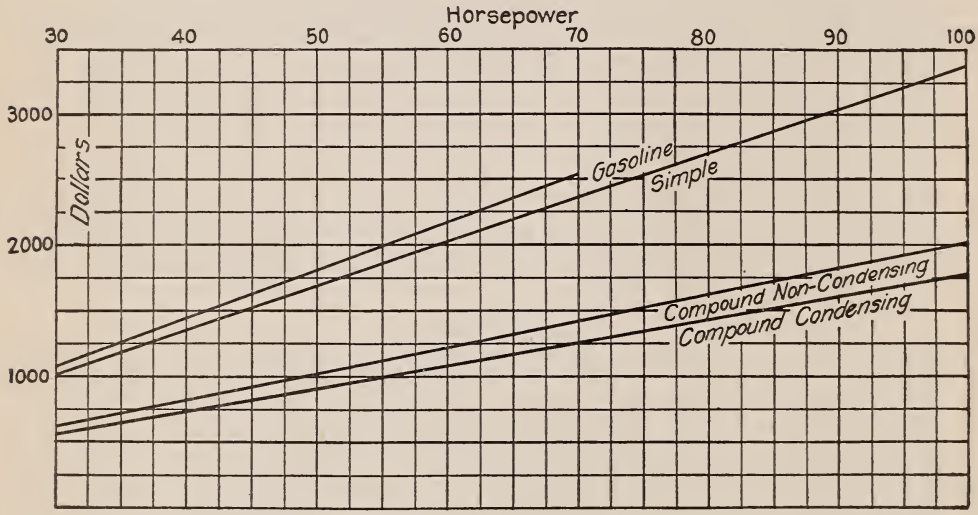

FIG. 19.—Cost of fuel per plant—coal $6 per ton, or gasoline 15 cents per gallon.

horsepower is 30. The simple steam engine and the gasoline engine are nearly equal under the conditions imposed, while the advantage is in favor of the compound engines, the condensing engine being the best proposition always, so far as cost of operation is concerned. When the price of fuel is increased 50 per cent the advantage of the compound condensing engine is more apparent.

Compound engines of 30 brake horsepower or less are built by few firms, while many firms build engines of 50 or more brake horsepower. The economy of operating the compound engines is sufficient to offset its increased cost. This is due in part to the use of a smaller and cheaper boiler than for the simple engine. Computations show that the price of coal would need to be about $1 per ton to make the cost of operating a simple steam engine and boiler as economical as operating a compound engine.

With fixed charges and labor increased the conditions would be changed slightly, but the fact remains that with plants of 30 brake horsepower or more the most economical engine to install is the compound condensing. For the smaller sizes, local or special conditions may affect the problem, but the choice will usually be between the gasoline and simple steam engines. Above 50 brake horsepower the choice is decidedly in favor of the compound engine and the higher the cost of fuel the more pronounced will be the gain. The slight increase of cost of the compound condensing engine and the possible increase in care in operating are well repaid in reduced fuel bills.

FOUNDATIONS FOR ENGINES.

Engines should be placed on foundations of ample size to prevent vibration and to give rigidity to the engine frame. Bricks or concrete are the materials employed for foundations usually. Engine builders furnish blueprints showing dimensions of the foundations they have found suitable for their engines. Where an outboard bearing is used, a concrete base with the support for the outboard bearing connected to the main foundation, the whole forming a monolith, makes a reliable support for an engine. Settling of the foundation in this case will not throw the engine out of line. Piles must be used when necessary to prevent excessive settling.

BOILERS.

Regardless of whether an economical or a wasteful steam engine is selected, a boiler is required to furnish the steam. The selection of a boiler of suitable size has been discussed. The size is usually expressed in boiler horsepower. The standard boiler horsepower is based on the transfer of a certain amount of heat per unit of time and usually computed on the basis of 10 or 12 square feet of heating surface per horsepower. It is equal to the evaporation of 34.5 pounds of water per hour from a feed-water temperature of 212° F. into steam at atmospheric pressure. This is approximately equal to 33,300 heat units per hour and, as stated above, 10 to 12 square feet, preferably the latter, of heating surface in a boiler with water on one side and the hot furnace gases on the other, is sufficient for this heat transfer.

TYPES OF BOILERS.

Water-tube boilers are seldom made in sizes under 50 horsepower. They require more attention in general than do the fire-tubular boilers, as the amount of water contained is relatively less, and they are therefore more sensitive to fluctuations of feed-water supply. The price of the smaller sizes of water-tube boilers is considerably in excess of the other type.

Some form of horizontal return-tubular boiler is commonly used in the small pumping plants. A semiportable boiler of the locomotive type is often used, which is usually on skids and requires little in the way of a foundation. These boilers, having internal furnaces, may be purchased in sizes up to 100 horsepower.

A cylindrical form of return-tubular boiler with brick setting or steel setting lined with fire bricks is often used also. This type of boiler is made in sizes ranging from 3 feet in diameter by 8 feet in length up to 8 feet in diameter by 24 feet in length. The number and size of tubes depend on the kind of fuel to be used and on the quality of the feed water.

Both types described are made in quantity and carried in stock by dealers. They cost less than boilers specially ordered, and when properly installed and operated intelligently are efficient and satisfactory.

SETTING FOR BOILERS.

The locomotive type of boiler requires no special setting, as the wooden skids serve this purpose. They should, however, be leveled up and sufficient foundations provided to insure permanency of position.

When the capacity exceeds 50 horsepower a horizontal return-tubular boiler is often used. Two kinds of settings are commonly used with these boilers; one consists of ordinary brick walls with a fire-brick lining where the hottest gases come in contact with the walls. This type is shown in figure 20, which shows lugs riveted to the sides of the boiler to rest on the brick walls and so support the weight. This method of support is not as desirable as that shown in figure 21, which could be used with the brick setting. It is well worth while to make a double wall with an air space with this type of setting, as shown in figure 20. The inner and outer walls will then expand independently, preventing the cracking of the setting, which causes air leaks and a lowering of efficiency.

The other setting, figure 21, consists of a steel setting lined with fire brick, placed either on side or on end so that the thickness varies from 4.5 to 9 inches. The thicker wall is better on account of the smaller losses in radiation. A sheet of asbestos is often placed between the fire bricks and setting. This kind of setting is less likely to have excessive air leaks than the brick setting. The cost is fully as great as the brick setting and possibly 10 per cent more, depending somewhat upon the details of the two designs. In this method the boiler is supported independently of its walls. This is accomplished by means of four cast-iron columns arranged in two pairs, one near the front end of the boiler and the other near the back. The boiler is suspended by tension rods from I-beams, with appropriate spread-

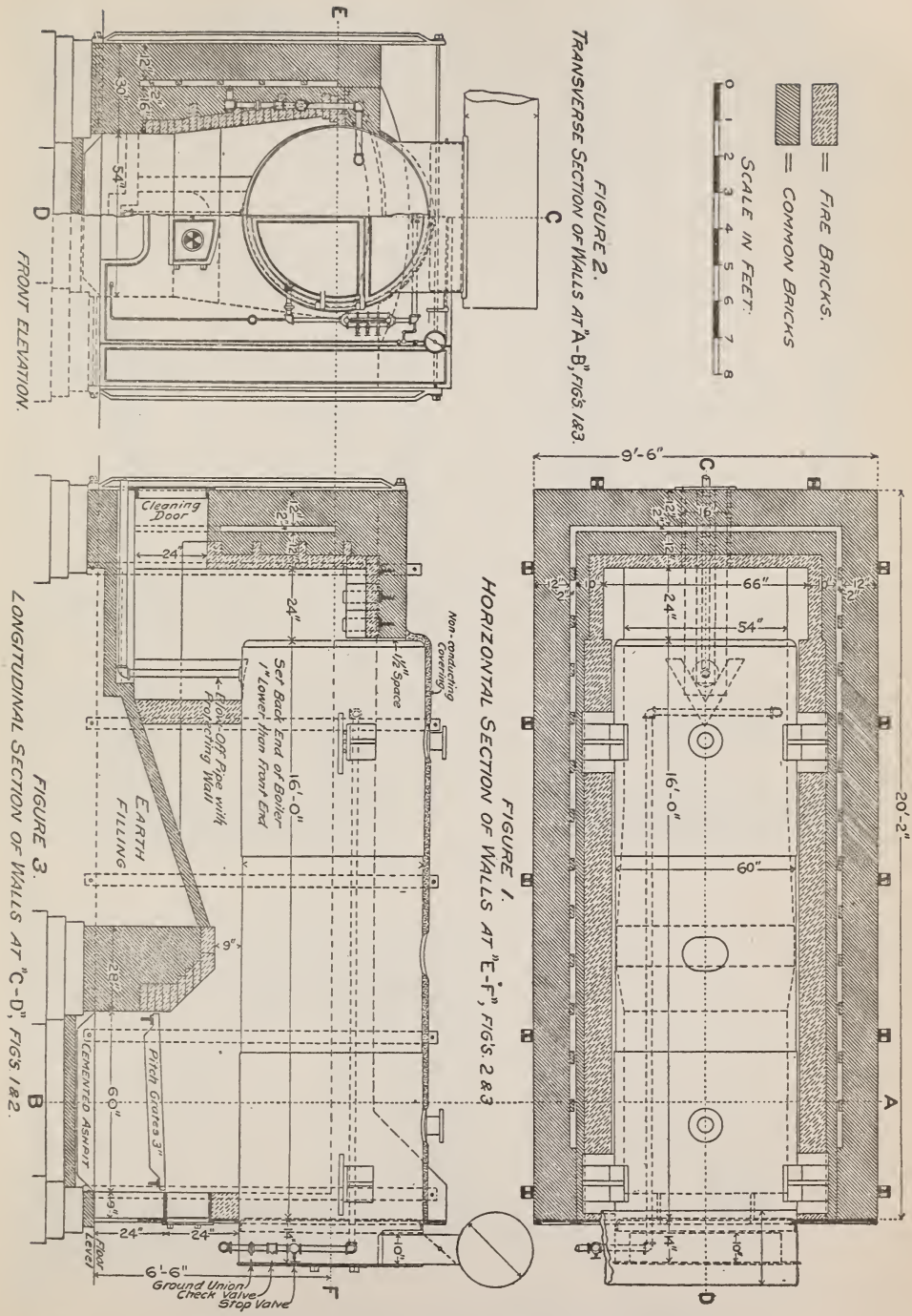

Fig. 20.—Brick boiler setting with fire-brick lining.

[Cir. 101]

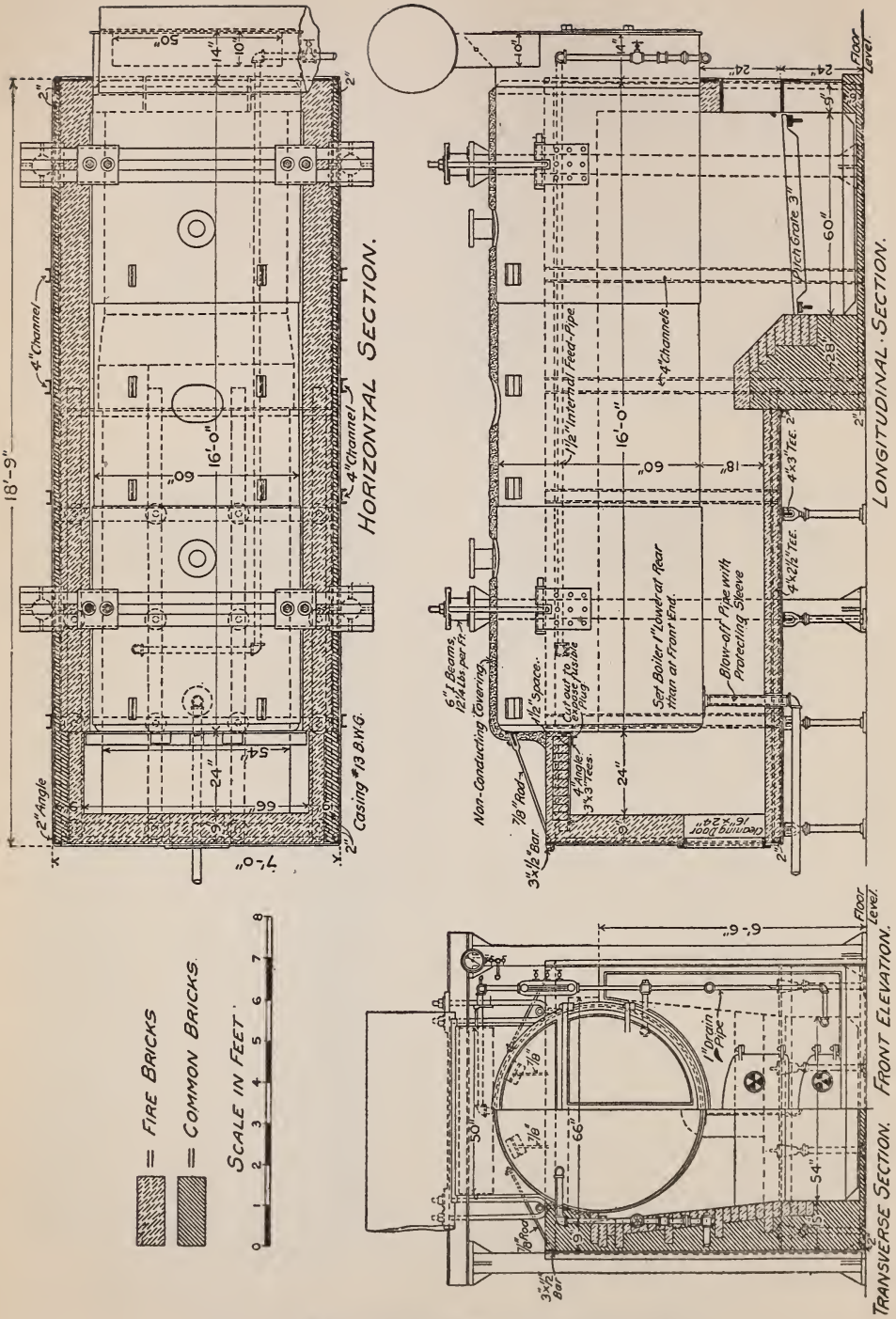

FIG. 21.—Steel boiler setting lined with fire brick.

[Cir. 101]

ers resting on each pair of columns. When the latter method of support is used in connection with the brick setting, wood is used sometimes as a cheap substitute for the cast-iron columns and steel beams. This is a bad practice because of the inflammability of the wood and also because of warping and lack of rigidity. While the first cost may be less for wood than for metal, if depreciation and repairs are considered over a term of years it will be found that the metal supports are cheaper. Whatever setting is used it should be designed so that it will be possible to inspect the boiler thoroughly, and particularly those parts that come in contact with the hottest gases. It is important also that the exposed portions of boilers be lagged. The lagging in the locomotive type should extend from the smoke box in front to the rear end of the boiler. The cost of a proper lagging for boilers and steam pipe may be offset by the saving of fuel in a single season. The top of a return-tubular boiler should be covered with some suitable material. The boiler and all its accessories should be protected from moisture. A small leak in the roof of a pump house, if allowed to continue through the season when the boiler is not in use, may cause corrosion that will shorten the life of a boiler materially. The smokestack should be well supported by steel guy lines securely anchored, and should be well painted to protect it from corrosion. Some kind of covering should be provided for the top when the stack is not in use, to prevent rain and moisture from injuring the interior and running down to corrode the breeching and boiler. A little expense and trouble in attending to these matters will add materially to the length of life of a boiler installation.

BOILER FITTINGS.

The blow-off pipe, where it passes downward near the rear of the boiler, must be protected from the hot gases, as the water does not circulate in the blow-off pipe. This is done either by means of a sleeve of larger-sized wrought-iron pipe or of tile or of brick. Failure to protect this pipe may lead to serious results, as an unprotected pipe may become so burned as to yield and cause an explosion. It is best to have two valves on the blow-off pipe so one may be closed and trouble prevented in case of accident to the other. A plug valve and a gate valve, placed close together, are often used.

The boiler should have a safety valve large enough to keep the pressure below a certain desired amount; either a weighted or a spring safety valve will answer the purpose. Means should be provided to try this valve every day so that it may be known definitely that it is not sticking to the valve seat.

The water column and pressure gauge should be of high-grade material, accurate, and reliable. It is well to connect the water

column so that there is no possibility of getting a wrong indication of water level through the accidental or careless closing of a valve. A pipe and valve should be provided to blow down the water from the bottom of the water column at short intervals.

Either a small steam pump or an injector is used to feed the boiler. A duplex pump, in which the valves on either side are moved by the action of the pistons on the opposite side, is a favorite for this work. The advantage of a pump over an injector is that the pump may be run continuously, its speed being determined by the opening of its steam valve, while an injector is operated usually only intermittently, unless it is exactly suited in capacity to the demands of the boilers. If a pump is used, it is well to have an injector also to feed the boiler in case of accident to the feed pump.

The feed-water pipe should be carried under the hot water in the boiler and for some distance before discharging its water. This allows the feed water to be heated to approximately the temperature of the water in the boiler and prevents bad temperature stresses in case some of the feed water, on leaving the pipe, comes into contact with the hot surfaces of the boiler shell.

HEATERS.

Some form of closed heater should be used in a steam plant unless the water contains impurities that will cause a deposit in the pipes. Where the water is bad and a deposit results, an open heater should be used, as it is much better to have the scale-forming substances taken out in the heater than deposited on the heating surface of the boiler. Fortunately, most of the troublesome scale-forming substances are precipitated by heat and the boiler may thus be protected by using a heater. The main reason for using a heater where the water is good is to heat the boiler feed by means of exhaust steam and so save heat that would be lost otherwise. A simple noncondensing engine converts a small percentage only of the heat of the steam into work and the exhaust steam contains 90 to 95 per cent of the heat that was in the steam as it came from the boiler to the engine. The temperature of the exhaust steam is approximately 212° F. for a noncondensing engine, while that for a condensing engine will be lower, the temperature depending upon the vacuum carried by the condenser. When an injector is used the heat taken from the boiler in the steam is returned to the boiler. When a feed pump is used, its exhaust may be turned into the heater and the greater part of the heat in the exhaust steam from the pump recovered. If a jet condenser and vacuum pump is used, the exhaust steam from this pump may also be utilized in the heater. The amount of steam from these two pumps is not enough to raise the feed

water to as high a temperature as in the case of the noncondensing engine, where the exhaust from the main engine is passed through the heater, but, on the other hand, the amount of water to be heated in a condensing plant is much smaller. It was assumed that the feed-water temperature in the noncondensing plant would be 190° F. and for the condensing plant 140° F.

The saving resulting from the use of a heater will range from 8 to 12 per cent of the steam and consequently of the fuel. The accompanying diagram (fig. 22) shows how much saving will result in a given case. The method of using the diagram is illustrated by the dotted lines. Given a temperature of water of 70° F., the saving resulting from raising its temperature in a heater to 200° F. is found at the left-hand side to be about 11.4 per cent. Again, if the tem-

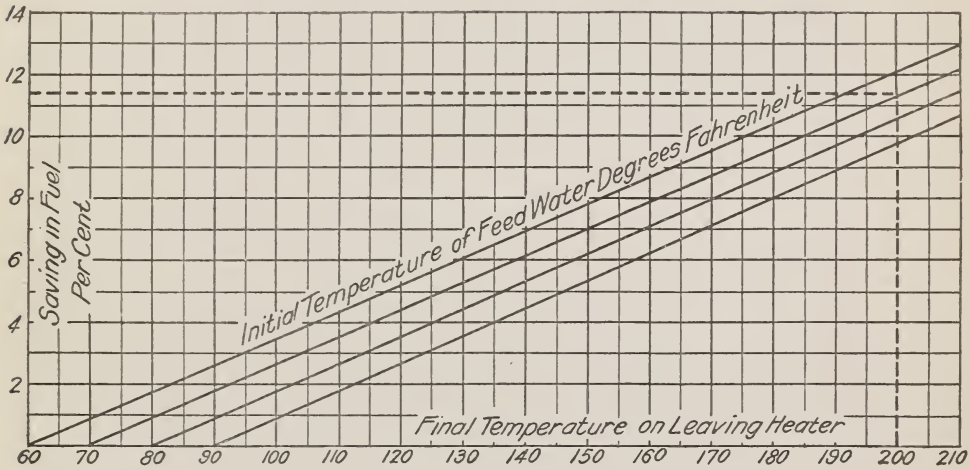

FIG. 22.—Saving of fuel resulting from use of water heater.

perature of the water before entering the heater is 60° F. and the temperature after leaving the heater is 175° F. the diagram shows a saving of 10 per cent in steam and fuel.

SUMMARY.

The points to be remembered in installing a small pumping plant may be briefly summarized as follows:

(1) Provide a pumping plant of proper capacity, selecting the various parts in sizes that will agree with each other, and therefore operate satisfactorily.

(2) Analyze the problem by the method shown and install the machinery that will produce the highest financial economy.

(3) Have the suction and discharge pipes of your pump of ample size, and enlarged at the ends if possible.

(4) If the power required is small and careful analysis shows that a gasoline engine is the best proposition, choose one that is well made

and that has a good reputation. Little details in design are largely accountable for the difference in reliability of operation of gasoline engines.

(5) If local conditions and size of pumping plant render the steam engine more economical, choose the type of engine that will make the cost of irrigating the least. In general, for a plant of capacity greater than 50 horsepower, the compound condensing engine should be used. Choose an engine of good material and workmanship.

(6) If a steam plant is selected, the boiler should be of ample size to furnish the necessary steam. The boiler and steam pipes should be thoroughly lagged. High-grade fittings are reliable, and in the end are cheaper.

(7) With a steam plant of any type use a feed-water heater. In a plant already erected and having a noncondensing engine a saving of from 8 to 12 per cent of the fuel will result from the installation of a feed-water heater.

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